



Technical Report

ISO/TR 10300-30

Calculation of load capacity of bevel gears —

Part 30: ISO rating system for bevel and hypoid gears — Sample calculations

*Calcul de la capacité de charge des engrenages coniques —
Partie 30: Système d'évaluation ISO pour engrenages conique et
hypoïde — Type de calculs*

**Second edition
2024-09**



COPYRIGHT PROTECTED DOCUMENT

© ISO 2024

All rights reserved. Unless otherwise specified, or required in the context of its implementation, no part of this publication may be reproduced or utilized otherwise in any form or by any means, electronic or mechanical, including photocopying, or posting on the internet or an intranet, without prior written permission. Permission can be requested from either ISO at the address below or ISO's member body in the country of the requester.

ISO copyright office
CP 401 • Ch. de Blandonnet 8
CH-1214 Vernier, Geneva
Phone: +41 22 749 01 11
Email: copyright@iso.org
Website: www.iso.org

Published in Switzerland

Contents

Foreword.....	v
Introduction.....	vi
1 Scope.....	1
2 Normative references.....	1
3 Terms and definitions	1
4 Symbols and abbreviated terms	1
5 Application	9
5.1 General.....	9
5.2 Structure of calculation methods.....	10
Annex A (informative) Sample 1: Rating of a spiral bevel gear pair without hypoid offset according to Method B1 and Method B2	12
A.1 Initial data	12
A.2 Calculation of Sample 1 according to Method B1	16
A.3 Calculation of Sample 1 according to Method B2	39
A.4 Calculation of Sample 1 according to local calculation method for surface durability – Method B1 localised.....	55
Annex B (informative) Sample 2: Rating of a hypoid gear set according to Method B1 and Method B2	93
B.1 Initial data	93
B.2 Calculation of Sample 2 according to Method B1	97
B.3 Calculation of Sample 2 according to Method B2	121
B.4 Calculation of Sample 2 according to local calculation method for surface durability – Method B1 localised.....	150
Annex C (informative) Sample 3: Rating of a hypoid gear set according to Method B1 and Method B2	182
C.1 Initial data	182
C.2 Calculation of Sample 3 according to Method B1	186
C.3 Calculation of Sample 3 according to Method B2	210
C.4 Calculation of Sample 3 according to local calculation method for surface durability – Method B1 localised.....	240
Annex D (informative) Sample 4: Rating of a hypoid gear set according to Method B1 and Method B2.....	272
D.1 Initial data	272
D.2 Calculation of Sample 4 according to Method B1	276
D.3 Calculation of Sample 4 according to Method B2	300
D.4 Calculation of Sample 4 according to local calculation method for surface durability – Method B1 localised.....	327

Annex E (informative) Graphical representation of the calculation results for Sample 1 to Sample 4	359
Bibliography	362

Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

The procedures used to develop this document and those intended for its further maintenance are described in the ISO/IEC Directives, Part 1. In particular, the different approval criteria needed for the different types of ISO documents should be noted. This document was drafted in accordance with the editorial rules of the ISO/IEC Directives, Part 2 (see www.iso.org/directives).

ISO draws attention to the possibility that the implementation of this document may involve the use of patents. ISO takes no position concerning the evidence, validity or applicability of any claimed patent rights in respect thereof. As of the date of publication of this document, ISO had not received notice of patents which may be required to implement this document. However, implementers are cautioned that this may not represent the latest information, which may be obtained from the patent database available at www.iso.org/patents. ISO shall not be held responsible for identifying any or all such patent rights.

Any trade name used in this document is information given for the convenience of users and does not constitute an endorsement.

For an explanation of the voluntary nature of standards, the meaning of ISO specific terms and expressions related to conformity assessment, as well as information about ISO's adherence to the World Trade Organization (WTO) principles in the Technical Barriers to Trade (TBT), see www.iso.org/iso/foreword.html.

This document was prepared by Technical Committee ISO/TC 60, *Gears*, Subcommittee SC 2, *Gear capacity calculation*.

This second edition cancels and replaces the first edition (ISO/TR 10300-30:2017), which has been technically revised.

The main changes are as follows:

- all sample calculations have been revised according to ISO 10300-1:2023, ISO 10300-2:2023 and ISO 10300-3:2023;
- all sample calculations include localised calculation method for Method B1.

A list of all parts in the ISO 10300 series can be found on the ISO website.

Any feedback or questions on this document should be directed to the user's national standards body. A complete listing of these bodies can be found at www.iso.org/members.html.

Introduction

The ISO 10300 series consists of International Standards, Technical Specifications (TS) and Technical Reports (TR) under the general title *Calculation of load capacity of bevel gears* (see Table 1).

- International Standards contain calculation methods that are based on widely accepted practices and have been validated.
- TS contain calculation methods that are still subject to further development.
- TR contain data that is informative, such as example calculations.

The procedures specified in ISO 10300-1 to ISO 10300-19 cover fatigue analyses for gear rating. The procedures described in ISO 10300-20 to ISO 10300-29 are predominantly related to the tribological behaviour of the lubricated flank surface contact. ISO 10300-30 to ISO 10300-39 include example calculations. The ISO 10300 series allows the addition of new parts under appropriate numbers to reflect knowledge gained in the future.

Requesting standardized calculations according to the ISO 10300 series without referring to specific parts requires the use of only those parts that are currently designated as International Standards (see Table 1 for listing). When requesting further calculations, the relevant part or parts of the ISO 10300 series need to be specified. The use of a Technical Specification as acceptance criteria for a specific design needs to be agreed in advance between manufacturer and purchaser.

Table 1 — Overview of the ISO 10300 series

Calculation of load capacity of bevel gears	International Standard	Technical Specification	Technical Report
<i>Part 1: Introduction and general influence factors</i>	X		
<i>Part 2: Calculation of surface durability (macropitting)</i>	X		
<i>Part 3: Calculation of tooth root strength</i>	X		
<i>Part 4 to 19: to be assigned</i>			
<i>Part 20: Calculation of scuffing load capacity – Flash temperature method</i>		X	
<i>Part 21 to 29: to be assigned</i>			
<i>Part 30: ISO rating system for bevel and hypoid gears — Sample calculations</i>			X
<i>Part 32: ISO rating system for bevel and hypoid gears — Sample calculation for scuffing load capacity</i>			X
NOTE At the time of publication of this document, some of the parts listed here were under development. Consult the ISO website.			

This document was prepared with sample calculations for different bevel gear designs. They are intended for users of the ISO 10300 series to follow a whole calculation procedure formula by formula. Practical experience has shown that this way, to get into a complex subject, is very helpful.

However, this document is not intended for use by the average engineer. Rather, it is aimed at the well-versed engineer capable of selecting reasonable values for the parameters and factors in these formulae based on knowledge of similar designs and on awareness of the effects behind these formulae.

Calculation of load capacity of bevel gears —

Part 30:

ISO rating system for bevel and hypoid gears — Sample calculations

1 Scope

This document provides sample calculations for the load capacity of different bevel gear designs, determined according to the methods and formulae of the ISO 10300 series. The initial geometric gear data necessary for these calculations are according to ISO 23509.

The term “bevel gear” is used to mean straight, helical (skew), spiral bevel, zerol and hypoid gear designs. Where this document pertains to one or more, but not all, the specific forms are identified.

The manufacturing process of forming the desired tooth form is not intended to imply any specific process, but rather to be general in nature and applicable to all calculation methods of the ISO 10300 series. The fact that there are bevel gear designs with tapered teeth and others where the tooth depth remains constant along the facewidth (uniform depth) does not require to apply Method B2 for the first and Method B1 for the second tooth configuration.

The rating system of the ISO 10300 series is based on virtual cylindrical gears and restricted to bevel gears whose virtual cylindrical gears have transverse contact ratios of $\varepsilon_{v\alpha} < 2$. Additionally, the given relations are valid for bevel gears of which the sum of profile shift coefficients of pinion and wheel is zero (see ISO 23509).

WARNING The user is cautioned that when the formulae are used for large average mean spiral angles, $(\beta_{m1} + \beta_{m2})/2 > 45^\circ$, for effective pressure angles, $\alpha_e > 30^\circ$ and/or for large facewidths, $b > 13 m_{mn}$, to confirm the calculated results of the ISO 10300 series by experience.

2 Normative references

The following documents are referred to in the text in such a way that some or all of their content constitutes requirements of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 10300-1:2023, *Calculation of load capacity of bevel gears — Part 1: Introduction and general influence factors*

ISO 10300-2:2023, *Calculation of load capacity of bevel gears — Part 2: Calculation of surface durability (macropitting)*

ISO 10300-3:2023, *Calculation of load capacity of bevel gears — Part 3: Calculation of tooth root strength*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 10300-1, ISO 10300-2 and ISO 10300-3 apply.

ISO and IEC maintain terminology databases for use in standardization at the following addresses:

— ISO Online browsing platform: available at <https://www.iso.org/obp>

— IEC Electropedia: available at <https://www.electropedia.org/>

4 Symbols and abbreviated terms

For the purposes of this document, the symbols and units given in ISO 10300-1:2023, Table 1 and Table 2, ISO 10300-2:2023, Table 1 and Table 2, ISO 10300-3:2023, Table 1 and Table 2 as well as the abbreviated terms given in ISO 10300-2:2023, Table 3, and ISO 10300-3:2023, Table 3, apply.

Table 2 — Symbols and units used in ISO 10300 (all parts)

Symbol	Description or term	Unit
A	Auxiliary factor for calculating the dynamic factor $K_v - c$	—
A^*	Related area for calculating the load sharing factor Z_{LS}	mm
A_{sne}	Outer tooth thickness allowance	mm
a	Hypoid offset	mm
a_{rel}	Relative hypoid offset	—
a_v	Centre distance of virtual cylindrical gear pair	mm
a_{vn}	Centre distance of virtual cylindrical gear pair in normal section	mm
B	Accuracy grade according to ISO 17485	—
b	Facewidth	mm
b_b	Related base facewidth	—
b_{ce}	Calculated effective facewidth	mm
b_{eff}	Effective facewidth (e.g. measured length of contact pattern)	mm
b_v	Facewidth of virtual cylindrical gears	mm
$b_{v,eff}$	Effective facewidth of virtual cylindrical gears	mm
C_F	Correction factor of tooth stiffness for non-average conditions	—
C_{lb}	Correction factor for the length of contact lines	—
C_{ZL}, C_{ZR}, C_{ZV}	Constants for determining lubricant film factors	—
C_{ham}	Mean addendum factor of wheel	—
C_v	Empirical parameter to determine the dynamic factor	—
c_γ	Mean value of mesh stiffness per unit facewidth	N/(mm · μ m)
$c_{\gamma 0}$	Mesh stiffness for average conditions	N/(mm · μ m)
c'	Single stiffness	N/(mm · μ m)
c_0'	Single stiffness for average conditions	N/(mm · μ m)
d_e	Outer pitch diameter	mm
d_m	Mean pitch diameter	mm
d_T	Tolerance diameter according to ISO 17485	mm
d_v	Reference diameter of virtual cylindrical gear	mm
d_{va}	Tip diameter of virtual cylindrical gear	mm
d_{van}	Tip diameter of virtual cylindrical gear in normal section	mm
d_{vb}	Base diameter of virtual cylindrical gear	mm

Symbol	Description or term	Unit
d_{vbn}	Base diameter of virtual cylindrical gear in normal section	mm
d_{vf}	Root diameter of virtual cylindrical gear	mm
d_{vn}	Reference diameter of virtual cylindrical gear in normal section	mm
E	Modulus of elasticity, Young's modulus	N/mm ²
E, G, H	Auxiliary variables for tooth form factor (Method B1)	—
e_{LS}	Exponent for the distribution of the load peaks along the lines of contact	—
F	Auxiliary variable for mid-zone factor	—
F_{mt}	Nominal tangential force at mid facewidth of the reference cone	N
F_{mtH}	Determinant tangential force at mid facewidth of the reference cone	N
F_n	Nominal normal force	N
F_{vmt}	Nominal tangential force of virtual cylindrical gears	N
f	Distance from the centre of the zone of action to a contact line	mm
f_{max}	Maximum distance to middle contact line	mm
f_{maxB}	Maximum distance to middle contact line at right side of the contact pattern	mm
f_{max0}	Maximum distance to middle contact line at left side of the contact pattern	mm
f_{pt}	Single pitch deviation	µm
$f_{p\ eff}$	Effective pitch deviation	µm
$f_{\alpha lim}$	Influence factor of limit pressure angle	
g_c	Length of contact line (Method B2)	mm
$g_{v\alpha}$	Length of path of contact of virtual cylindrical gear in transverse section	mm
$g_{v\alpha n}$	Related length of action in normal section	—
g_l	Length of action from mean point to point of load application (Method B2)	mm
g_η	Relative length of action within the contact ellipse	mm
HB	Brinell hardness	—
h_{am}	Mean addendum	mm
h_{a0}	Tool addendum	mm
h_{Fa}	Bending moment arm for tooth root stress (load application at tooth tip)	mm
h_{fm}	Mean dedendum	mm
h_{fP}	Dedendum of the basic rack profile	mm
h_m	Mean whole depth used for bevel spiral angle factor	mm
h_{vfm}	Relative mean virtual dedendum	—
h_N	Load height from critical section (Method B2)	mm
j_{en}	Outer normal backlash	mm

ISO/TR 10300-30:2024(E)

Symbol	Description or term	Unit
K	Constant; factor for calculating the dynamic factor K_{v-B}	—
K_A	Application factor	—
K_{F0}	Lengthwise curvature factor for bending stress	—
$K_{F\alpha}$	Transverse load factor for bending stress	—
$K_{F\alpha}^*$	Preliminary transverse load factor for bending stress for non-hypoid gears	—
$K_{F\beta}$	Face load factor for bending stress	—
$K_{H\alpha}$	Transverse load factor for contact stress	—
$K_{H\alpha}^*$	Preliminary transverse load factor for contact stress for non-hypoid gears	—
$K_{H\beta}$	Face load factor for contact stress	—
$K_{H\beta-be}$	Mounting factor	—
K_v	Dynamic factor	—
K_v^*	Preliminary dynamic factor for non-hypoid gears	—
k'	Contact shift factor	—
k_c	Clearance factor	—
k_d	Depth factor	—
k_{hap}	Basic crown gear addendum factor (related to m_{mn})	—
k_{hfp}	Basic crown gear dedendum factor (related to m_{mn})	—
k_t	Circular thickness factor	—
l_b	Length of contact line (Method B1)	mm
l_{b0}	Theoretical length of contact line	mm
l_{bm}	Theoretical length of middle contact line	mm
m_{et}	Outer transverse module	mm
m_{mn}	Mean normal module	mm
m_{mt}	Mean transverse module	mm
m_{red}	Mass per unit facewidth reduced to the line of action of dynamically equivalent cylindrical gears	kg/mm
m^*	Related individual gear mass per unit facewidth referred to the line of action	kg/mm
N	Reference speed related to resonance speed n_{E1}	—
N_L	Number of load cycles	—
n	Rotational speed	min ⁻¹
n_{E1}	Resonance speed of pinion	min ⁻¹
P	Nominal power	kW
p	Peak load	N/mm
p_{et}	Transverse base pitch (Method B2)	mm
p_{max}	Maximum peak load	N/mm
p^*	Related peak load for calculating the load sharing factor (Method B1)	—

Symbol	Description or term	Unit
p_{mn}	Relative mean normal pitch	—
p_{nb}	Relative mean normal base pitch	—
p_{vet}	Transverse base pitch of virtual cylindrical gear (Method B1)	mm
q	Exponent in the formula for lengthwise curvature factor	—
q_s	Notch parameter	—
q_s	Notch parameter (Method B2)	—
Ra	= CLA = AA arithmetic average roughness	µm
R_e	Outer cone distance	mm
R_m	Mean cone distance	mm
R_{mpt}	Relative mean back cone distance	—
Rz	Mean roughness	µm
RZ_{10}	Mean roughness for gear pairs with relative curvature radius $\rho_{rel} = 10 \text{ mm}$	µm
r_{c0}	Cutter radius	mm
r_{mf}	Tooth fillet radius at the root in mean section	mm
r_{mpt}	Mean pitch radius	mm
$r_{my 0}$	Mean transverse radius to point of load application (Method B2)	mm
r_{va}	Relative mean virtual tip radius	—
r_{vn}	Relative mean virtual pitch radius	—
S_F	Safety factor for bending stress (against breakage)	—
$S_{F \min}$	Minimum safety factor for bending stress	—
S_H	Safety factor for contact stress (against macropitting)	—
$S_{H \min}$	Minimum safety factor for contact stress	—
S_{Fn}	Tooth root chord in calculation section	mm
s_{mn}	Mean normal circular thickness	mm
s_N	One-half tooth thickness at critical section (Method B2)	mm
s_{pr}	Amount of protuberance at the tool	mm
$T_{1,2}$	Nominal torque of pinion and wheel	Nm
u	Gear ratio of bevel gear	—
u_v	Gear ratio of virtual cylindrical gear	—
v_{et}	Tangential speed at outer end (heel) of the reference cone	m/s
$v_{et \max}$	Maximum pitch line velocity at operating pitch diameter	m/s
v_g	Sliding velocity in the mean point P	m/s
$v_{g \text{ par}}$	Sliding velocity parallel to the contact line	m/s
$v_{g \text{ vert}}$	Sliding velocity vertical to the contact line	m/s
v_{mt}	Tangential speed at mid facewidth of the reference cone	m/s
v_{Σ}	Sum of velocities in the mean point P	m/s

Symbol	Description or term	Unit
$v_{\Sigma h}$	Sum of velocities in profile direction	m/s
$v_{\Sigma l}$	Sum of velocities in lengthwise direction	m/s
$v_{\Sigma \text{ vert}}$	Sum of velocities vertical to the contact line	m/s
W_{m2}	Wheel mean slot width	mm
w	Angle of contact line relative to the root cone	°
x_{hm}	Profile shift coefficient	—
x_{sm}	Thickness modification coefficient (backlash included)	—
x_{smn}	Thickness modification coefficient (theoretical)	—
x_N	Tooth strength factor (Method B2)	mm
x_{oo}	Distance from mean section to point of load application	mm
Y_A	Root stress adjustment factor (Method B2)	—
Y_{BS}	Bevel spiral angle factor	—
Y_{Fa}	Tooth form factor for load application at the tooth tip (Method B1)	—
Y_{FS}	Combined tooth form factor for generated gears	—
Y_f	Stress concentration and stress correction factor (Method B2)	—
Y_i	Inertia factor (bending)	—
Y_j	Bending strength geometry factor (Method B2)	—
Y_{LS}	Load sharing factor (bending)	—
Y_{NT}	Life factor (bending)	—
$Y_{R \text{ rel T}}$	Relative surface condition factor	—
Y_{Sa}	Stress correction factor for load application at the tooth tip	—
Y_{ST}	Stress correction factor for dimensions of the standard test gear	—
Y_X	Size factor for tooth root stress	—
$Y_{1,2}$	Tooth form factor of pinion and wheel (Method B2)	—
$Y_{\delta \text{ rel T}}$	Relative notch sensitivity factor	—
Y_{ϵ}	Contact ratio factor for bending (Method B1)	—
y_p	Running-in allowance for pitch deviation related to the polished test piece	µm
y_j	Location of point of load application for maximum bending stress on path of action (Method B2)	mm
y_3	Location of point of load application on path of action for maximum root stress	mm
y_{α}	Running-in allowance for pitch error	µm
Z_A	Contact stress adjustment factor (Method B2)	—
Z_E	Elasticity factor	—
Z_{FW}	Facewidth factor	—
Z_{Hyp}	Hypoid factor	—

Symbol	Description or term	Unit
Z_I	Macropitting resistance geometry factor (Method B2)	—
Z_i	Inertia factor (macropitting)	—
Z_{KP}	Bevel gear factor (Method B1)	—
Z_L	Lubricant factor	—
Z_{LS}	Load sharing factor (Method B1)	—
Z_{M-B}	Mid zone factor	—
Z_{NT}	Life factor (macropitting)	—
Z_R	Roughness factor for contact stress	—
Z_S	Bevel slip factor	—
Z_v	Speed factor	—
Z_W	Work hardening factor	—
Z_X	Size factor	—
z	Number of teeth	—
z_v	Number of teeth of virtual cylindrical gear	—
z_{vn}	Number of teeth of virtual cylindrical gear in normal section	—
z_0	Number of blade groups of the cutter	—
α_a	Adjusted pressure angle (Method B2)	°
α_{an}	Normal pressure angle at tooth tip	°
$\alpha_{dD,C}$	Nominal design pressure angle for drive side/coast side	°
α_{et}	Effective pressure angle in transverse section	°
$\alpha_{eD,C}$	Effective pressure angle for drive side/coast side	°
α_f	Limit pressure angle in wheel root coordinates (Method B2)	°
α_{lim}	Limit pressure angle	°
$\alpha_{nD,C}$	Generated pressure angle for drive side/coast side	°
α_{vet}	Transverse pressure angle of virtual cylindrical gears	°
α_{Fan}	Load application angle at tooth tip of virtual cylindrical gear (Method B1)	°
α_L	Normal pressure angle at point of load application (Method B2)	°
β_{bm}	Mean base spiral angle	°
β_m	Mean spiral angle	°
β_v	Helix angle of virtual gear (Method B1), virtual spiral angle (Method B2)	°
β_{vb}	Helix angle at base circle of virtual cylindrical gear	°
β_B	Inclination angle of contact line	°
γ	Auxiliary angle for length of contact line calculation (Method B1)	°
γ'	Projected auxiliary angle for length of contact line	°
γ_a	Auxiliary angle for tooth form and tooth correction factor	°

Symbol	Description or term	Unit
δ	Pitch angle of bevel gear	°
δ_a	Face angle	°
δ_f	Root angle	°
$\varepsilon_{v\alpha}$	Transverse contact ratio of virtual cylindrical gears	—
$\varepsilon_{v\alpha n}$	Transverse contact ratio of virtual cylindrical gears in normal section	—
$\varepsilon_{v\beta}$	Face contact ratio of virtual cylindrical gears	—
$\varepsilon_{v\gamma}$	Virtual contact ratio (Method B1), modified contact ratio (Method B2)	—
ε_N	Load sharing ratio for bending (Method B2)	—
ε_{NI}	Load sharing ratio for macropitting (Method B2)	—
ζ_m	Pinion offset angle in axial plane	°
ζ_{mp}	Pinion offset angle in pitch plane	°
ζ_R	Pinion offset angle in root plane	°
θ	Auxiliary quantity for tooth form and tooth correction factors	—
θ_{mp}	Auxiliary angle for virtual facewidth (Method B1)	°
θ_{a2}	Addendum angle of wheel	°
θ_{f2}	Dedendum angle of wheel	°
θ_{v2}	Angular pitch of virtual cylindrical wheel	radiant
ν	Poisson's ratio	—
ν_0	Lead angle of face hobbing cutter	°
ν_{40}, ν_{50}	Nominal kinematic viscosity of the oil at 40 °C and 50 °C, respectively	mm ² /s
ξ	Assumed angle in locating weakest section	°
ξ_h	One half of angle subtended by normal circular tooth thickness at point of load application	°
ρ	Density of gear material	kg/mm ³
ρ_{a0}	Cutter edge radius	mm
ρ_F	Fillet radius at point of contact of 30° tangent	mm
ρ_{Fn}	Fillet radius at point of contact of 30° tangent in normal section	mm
ρ_{fP}	Root fillet radius of basic rack for cylindrical gears	mm
ρ_{rel}	Radius of relative curvature vertical to contact line at virtual cylindrical gears	mm
ρ_t	Radius of relative profile curvature (Method B2)	mm
ρ_{va0}	Relative edge radius of tool	—
ρ'	Slip layer thickness	mm
Σ	Shaft angle	°
σ_F	Tooth root stress	N/mm ²
σ_{F0}	Nominal tooth root stress	N/mm ²
$\sigma_{F \lim}$	Nominal stress number (bending)	N/mm ²

Symbol	Description or term	Unit
σ_{FE}	Allowable stress number (bending)	N/mm ²
σ_{FP}	Permissible tooth root stress	N/mm ²
σ_H	Contact stress	N/mm ²
$\sigma_{H \text{ lim}}$	Allowable stress number for contact stress	N/mm ²
σ_{HP}	Permissible contact stress	N/mm ²
τ	Angle between tangent of root fillet at weakest point and centreline of tooth	°
ϕ	Auxiliary angle to determine the position of the pitch point	°
χ^x	Relative stress drop in notch root	mm ⁻¹
χ_T^x	Relative stress drop in notch root of standardized test gear	mm ⁻¹
ω	Angular velocity	rad/s
ω_Σ	Angle between the sum of velocities vector and the trace of pitch cone	°

Table 3 — Generally used subscripts in ISO 10300 (all parts)

Subscripts	Description
0	Tool
1	Pinion
2	Wheel
A, B, B1, B2, C	Value according to Method A, B, B1, B2 or C
D	Drive flank
C	Coast flank
T	Relative to standardized test gear dimensions
(1), (2)	Trials of interpolation

5 Application

5.1 General

This document provides four sample calculations:

- Sample 1 is a rating of a spiral bevel gear pair without hypoid offset according to Method B1, including the localised calculation method, and Method B2 (see Annex A);
- Sample 2 is a rating of a hypoid gear set according to Method B1, including the localised calculation method, and Method B2 (see Annex B);
- Sample 3 is a rating of a hypoid gear set according to Method B1, including the localised calculation method, and Method B2 (see Annex C);
- Sample 4 is a rating of a hypoid gear set according to Method B1, including the localised calculation method, and Method B2 (see Annex D).
- Figures E.1 to E.3 provide a graphical representation of the calculation results for Samples 1 to 4.

While this document states the results of calculations to a three decimal accuracy, the calculations themselves use more significant digits.

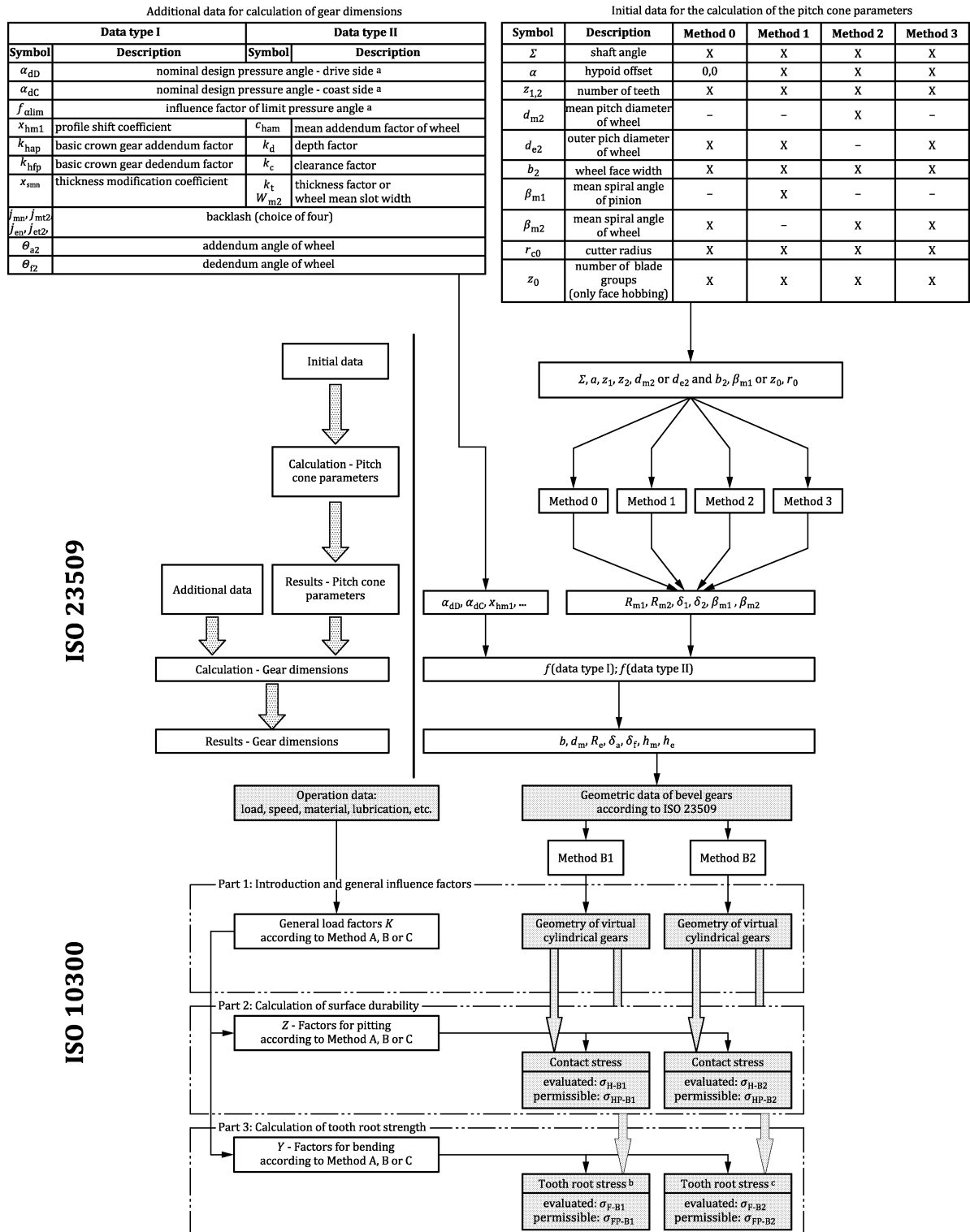
Table 4 refers to each table of the annexes showing the input parameters and calculation results for the different sample calculation steps.

Table 4 — Content of tables of Annexes A to D

Content of Table	Sample 1	Sample 2	Sample 3	Sample 4
Initial data for pitch cone parameters	Table A.1	Table B.1	Table C.1	Table D.1
Input data for tooth profile parameters	Table A.2	Table B.2	Table C.2	Table D.2
Geometric data from calculation according to ISO 23509	Table A.3	Table B.3	Table C.3	Table D.3
Operation parameters and additional considerations	Table A.4	Table B.4	Table C.4	Table D.4
Method B1				
Virtual cylindrical gears	Table A.5	Table B.5	Table C.5	Table D.5
General influence factors	Table A.6	Table B.6	Table C.6	Table D.6
Calculation of surface durability (macropitting)	Table A.7	Table B.7	Table C.7	Table D.7
Calculation of tooth root strength for pinion	Table A.8	Table B.8	Table C.8	Table D.8
Calculation of tooth root strength for wheel	Table A.9	Table B.9	Table C.9	Table D.9
Method B2				
Calculation of surface durability (macropitting)	Table A.10	Table B.10	Table C.10	Table D.10
Calculation of tooth root strength on pinion	Table A.11	—	—	—
Calculation of tooth root strength on pinion and wheel	—	Table B.11	Table C.11	Table D.11
Calculation of tooth root strength on wheel	Table A.12	—	—	—
Method B1 localised				
Virtual cylindrical gears	Table A.13	Table B.12	Table C.12	Table D.12
Local velocities and slip	Table A.14	Table B.13	Table C.13	Table D.13
Calculation of local surface durability (macropitting)	Table A.15	Table B.14	Table C.14	Table D.14

5.2 Structure of calculation methods

Figure 1 shows three boxes that represent the individual three parts of ISO 10300. However, these boxes are subdivided into a left side where influence factors are determined on the basis of mainly operational data according to Methods A, B or C (see ISO 10300-1:2023, 5.1) and a right side where separate calculation procedures are provided according to Method B1 and Method B2 which are assumed to have the same level B but different approaches. These two methods refer to the determination of virtual cylindrical gears in ISO 10300-1, the gear flank rating formulae in ISO 10300-2 and the gear tooth rating formulae in ISO 10300-3.



- ^a Generally drive and coast side pressure angles are balanced in initial design. However, some applications may be optimized with unbalanced pressure angles, see ISO 23059:2016, Annex C for guidance.
- ^b One set of formulae for both, bevel and hypoid gears.
- ^c Separate sets of formulae for bevel and for hypoid gears.

Figure 1 — Structure of calculation methods in ISO 10300 (all parts)

Annex A

(informative)

Sample 1: Rating of a spiral bevel gear pair without hypoid offset according to Method B1 and Method B2

A.1 Initial data

Sample 1 is for a spiral bevel gear pair without hypoid offset which uses Method 0 according to ISO 23509.

Table A.1 — Initial data for pitch cone parameters

Symbol	Description	Method 0	Method 1	Method 2	Method 3
Σ	Shaft angle	90°	X	X	X
a	Hypoid offset	0 mm	X	X	X
$z_{1,2}$	Number of teeth	14/39	X	X	X
d_{m2}	Mean pitch diameter of wheel	—	—	X	—
d_{e2}	Outer pitch diameter of wheel	176,893 mm	X	—	X
b_2	Wheel facewidth	25,4 mm	X	X	X
β_{m1}	Mean spiral angle of pinion	35°	X	—	—
β_{m2}	Mean spiral angle of wheel	35°	—	X	X
r_{c0}	Cutter radius	114,3 mm	X	X	X
z_0	Number of blade groups (only face hobbing)	—	—	X	X

Table A.2 — Input data for tooth profile parameters

Data type I		Data type II	
Symbol	Description	Symbol	Description
α_{dB}		20°	
α_{dC}		20°	
f_{alim}		0	
X_{hm1}	—	$Cham$	0,247 37
k_{hap}	—	k_d	2,000
k_{hfp}	—	k_c	0,125
X_{smn}	—	k_t	0,091 5
		W_{m2}	—
j_{en}		0,127 mm	
θ_{a2}		2,134 2°	
θ_{r2}		6,493 4°	
$\rho_{a01D,C}$		0,8 mm/0,8 mm	
$\rho_{a02D,C}$		1,2 mm/1,2 mm	
$S_{pr1D,C}$		0 mm/0 mm	
$S_{pr2D,C}$		0 mm/0 mm	

Table A.3 and Table A.4 show geometric and operational data and text for explanation.

Table A.3 — Geometric data from calculation according to ISO 23509

Symbol	Description	Values	Symbol	Description	Value
$d_{m1,2}$	Mean pitch diameter of pinion/wheel	54,918 mm/ 152,987 mm	ζ_{mp}	Offset angle on pitch plane	0°
$h_{am1,2}$	Mean addendum of pinion/wheel	4,837 mm/ 1,590 mm	ζ_R	Pinion offset angle on root plane	0°
$h_{fm1,2}$	Mean dedendum of pinion/wheel	2,393 mm/ 5,640 mm	$R_{e1,2}$	Outer cone distance on pinion and wheel	93,973 mm
$\alpha_{eD,C}$	Effective pressure angle for drive side/coast side	20°/20°	$R_{m1,2}$	Mean cone distance on pinion and wheel	81,273 mm
$\alpha_{nD,C}$	Generated pressure angle for drive side/coast side	20°/20°	$\delta_{1,2}$	Pitch angle on pinion/wheel	19,747°/ 70,253°
α_{im}	Limit pressure angle	0°	$\delta_{a1,2}$	Face angle on pinion/wheel	26,240°/ 72,387°
m_{mn}	Mean normal module	3,213 mm	$\delta_{f1,2}$	Root angle on pinion/wheel	17,613°/ 63,760°
k_{hfp}	Basic crown gear dedendum factor	1,25	$x_{sm1,2}$	Thickness modification coefficient on pinion/wheel	0,036/ -0,055
ζ_m	Pinion offset angle on axial plane	0,000°	m_{et2}	Outer transverse module	4,536 mm
$s_{mn1,2}$	Mean normal circular tooth thickness of pinion/wheel	6,464 mm/ 3,511 mm	—		

Table A.4 — Operation parameters and additional considerations

Symbol	Description	Value
Additional data		
	Wheel profile	Generated
	Roughing/finishing method	Face milling
$b_{2\text{eff}}$	Effective facewidth on wheel	$0,85 \cdot b_2$
	Profile crowning	Low
	Verification of contact pattern	Checked under light test load for each gear
	Mounting conditions of pinion and wheel	One member cantilever-mounted
Operation parameters		
T_1	Pinion torque	300 Nm
n_1	Pinion rotational speed	1 200 min ⁻¹
K_A	Application factor	1,1
	Active flank	Drive
Material data for pinion and wheel (case hardened steel)		
E	Modulus of elasticity	210 000 N/mm ²
ν	Poisson's ratio	0,3
$\sigma_{H \text{ lim}}$	Allowable stress number (contact)	1 500 N/mm ²
$\sigma_{F \text{ lim}}$	Nominal stress number (bending)	480 N/mm ²
	Surface hardness	Same for pinion and wheel
Quality parameters		
R_z	Flank roughness on pinion/wheel	8 µm/8 µm
R_z	Tooth root roughness on pinion/wheel	16 µm/16 µm
f_{pt}	Single pitch deviation on pinion/wheel	12 µm/26 µm
Lubrication parameters		
	Oil type	ISO-VG-150
	Oil temperature	90 °C

A.2 Calculation of Sample 1 according to Method B1

Table A.5 — Virtual cylindrical gears

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Virtual cylindrical gears in transverse section					
Reference diameter on pinion	$d_{v1} = \frac{d_{m1}}{\cos \delta_1}$	58,350 mm	E: A.1		
Reference diameter on wheel	$d_{v2} = u^2 \cdot d_{v1}$	452,804 mm	E: A.4		
Helix angle	$\beta_v = \frac{\beta_{m1} + \beta_{m2}}{2}$	35°	E: A.8		
Transverse pressure angle of virtual cylindrical gears	$\alpha_{vet} = \arctan(\tan \alpha_e / \cos \beta_v)$ since $\alpha_e = \alpha_{eD}$ for drive side	23,957°	E: A.10		
Transverse module	$m_{vt} = m_{mn} / \cos \beta_v$	3,923 mm	E: A.11		
Number of teeth on pinion	$z_{v1} = d_{v1} / m_{vt}$	14,875	E: A.12		
Number of teeth on wheel	$z_{v2} = d_{v2} / m_{vt}$	115,431	E: A.12		
Gear ratio	$u_v = z_{v2} / z_{v1}$	7,760	E: A.13		

ISO/TR 10300-30:2024(E)

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Auxiliary angle for virtual facewidth	$\vartheta_{mp} = \arctan(\sin \delta_2 \cdot \tan \zeta_m)$	0°	E: A.21		
Projected auxiliary angle for length of contact line	$\gamma' = \vartheta_{mp} - \zeta_{mp} / 2$	0°	E: A.20		
Centre distance of virtual cylindrical gear pair	$a_v = (d_{v1} + d_{v2}) / 2$	255,577 mm	E: A.5		
Helix angle of virtual cylindrical gear at base circle	$\beta_{vb} = \arcsin(\sin \beta_v \cdot \cos \alpha_e)$ since $\alpha_e = \alpha_{eD}$ for drive side	32,615°	E: A.16		
Tip diameter on pinion	$d_{va1} = d_{v1} + 2 h_{am1}$	68,023 mm	E: A.6		
Tip diameter on wheel	$d_{va2} = d_{v2} + 2 h_{am2}$	455,984 mm	E: A.6		
Root diameter on pinion	$d_{vf1} = d_{v1} - 2 h_{fm1}$	53,563 mm	E: A.7		
Root diameter on wheel	$d_{vf2} = d_{v2} - 2 h_{fm2}$	441,524 mm	E: A.7		
Base diameter on pinion	$d_{vb1} = d_{v1} \cdot \cos \alpha_{vet}$	53,323 mm	E: A.9		
Base diameter on wheel	$d_{vb2} = d_{v2} \cdot \cos \alpha_{vet}$	413,796 mm	E: A.9		
Transverse base pitch	$p_{vet} = \pi \cdot m_{mn} \cdot \cos \alpha_{vet} / \cos \beta_v$	11,262 mm	E: A.17		

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Length of path of contact of virtual cylindrical gear in transverse section	$g_{v\alpha} = \frac{1}{2} \cdot \left[\left(\sqrt{d_{va1}^2 - d_{vb1}^2} - d_{v1} \cdot \sin \alpha_{vet} \right) + \left(\sqrt{d_{va2}^2 - d_{vb2}^2} - d_{v2} \cdot \sin \alpha_{vet} \right) \right]$	13,120 mm	E: A.18		
Transverse contact ratio	$\varepsilon_{v\alpha} = g_{v\alpha} / p_{vet}$	1,165	E: A.23		
Effective facewidth with $b_{2\text{ eff}} = 0,85 \cdot b_2$	$b_{v,\text{eff}} = \frac{(b_{2,\text{eff}} / \cos(\zeta_{mp} / 2) - g_{v\alpha} \cdot \cos \alpha_{vet} \cdot \tan(\zeta_{mp} / 2))}{1 - \tan \gamma' \cdot \tan(\zeta_{mp} / 2)}$	21,590 mm	E: A.19		
Facewidth	$b_v = b_2 \cdot \frac{b_{v,\text{eff}}}{b_{2,\text{eff}}}$	25,400 mm	E: A.22		
Virtual cylindrical gears in normal section					
Number of pinion teeth of virtual cylindrical gears	$z_{vn1} = \frac{z_{v1}}{\cos^2 \beta_{vb} \cdot \cos \beta_v}$	25,594	E: A.41		
Number of wheel teeth of virtual cylindrical gears	$z_{vn2} = u_v \cdot z_{vn1}$	198,613	E: A.42		
Reference diameter on pinion	$d_{vn1} = z_{vn1} \cdot m_{mn}$	82,241 mm	E: A.43		
Reference diameter on wheel	$d_{vn2} = z_{vn2} \cdot m_{mn}$	638,207 mm	E: A.43		
Tip diameter on pinion	$d_{van1} = d_{vn1} + 2h_{am1}$	91,915 mm	E: A.44		
Tip diameter on wheel	$d_{van2} = d_{vn2} + 2h_{am2}$	641,387 mm	E: A.44		

ISO/TR 10300-30:2024(E)

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Base diameter on pinion	$d_{vbn1} = d_{vn1} \cdot \cos \alpha_e$	77,281 mm	E: A.45		
Base diameter on wheel	$d_{vbn2} = d_{vn2} \cdot \cos \alpha_e$	599,719 mm	E: A.45		
Face contact ratio	$\varepsilon_{v\beta} = \frac{b_{v,eff} \cdot \sin \beta_v}{\pi \cdot m_{mn}}$	1,227	E: A.24		
Virtual contact ratio	$\varepsilon_{v\gamma} = \varepsilon_{v\alpha} + \varepsilon_{v\beta}$	2,392	E: A.25		
Inclination angle of contact line	$\beta_B = \arctan(\tan \beta_v \cdot \sin \alpha_e)$	13,468°	E: A.38		
Radius of relative curvature in normal section at the mean point	$\rho_t = \left[\frac{1}{\cos \alpha_{nD} \cdot (\tan \alpha_{nD} - \tan \alpha_{lim}) + \tan \zeta_{mp} \cdot \tan \beta_B} \cdot \frac{\cos \beta_{m1} \cdot \cos \beta_{m2}}{\cos \zeta_{mp}} \cdot \left(\frac{1}{R_{m2} \cdot \tan \delta_2} + \frac{1}{R_{m1} \cdot \tan \delta_1} \right) \right]^{-1}$	13,173 mm	E: A.39		
Radius of relative curvature vertical to the contact line	$\rho_{rel} = \rho_t \cdot (\cos \beta_B)^2$	12,459 mm	E: A.37		

Table A.6 — General influence factors

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Nominal tangential force of bevel gears	$F_{mt1,2} = \frac{2000 \cdot T_{1,2}}{d_{m1,2}}$	10 925,3 N	E: 1		
Nominal tangential force of virtual cylindrical gears	$F_{vmt} = F_{mt1} \cdot \frac{\cos \beta_v}{\cos \beta_{m1}}$	10 925,3 N	E: 2		
Nominal tangential speed at mean point of the pinion	$v_{mt1} = \frac{d_{m1} \cdot n_1}{19\,098}$	3,451 m/s	E: 5		
Nominal tangential speed at mean point of the wheel	$v_{mt2} = \frac{d_{m2} \cdot n_2}{19\,098}$	3,451 m/s	E: 5		
Correction factor for non-average conditions for $F_{vmt} K_A / b_{v,eff} \geq 100$ N/mm	C_F	1,000	E: 12		
Mean value of mesh stiffness per unit face width	$c_\gamma = c_{\gamma 0} \cdot C_F$	20 N/(mm · µm)	E: 11		
Single stiffness	$c' = c'_0 \cdot C_F$	14 N/(mm · µm)	E: 18		
Max. single pitch deviation as given in Table 1.4.	f_{pt}	26 µm			

ISO/TR 10300-30:2024(E)

Full document details: ISO TR 10300-30-2024

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Running-in allowance for case hardened and nitrided gears	$y_{\alpha} = 0,075 \cdot f_{pt}$	1,950 μm	E: 51		
Effective pitch deviation with $y_p = y_{\alpha}$	$f_{p,eff} = f_{pt} - y_p$	24,050 μm	E: 17		
Relative pinion mass per unit facewidth reduced to the line of action	$m_1^* = \frac{1}{8} \cdot \rho \cdot \pi \cdot \frac{d_{m1}^2}{\cos^2[(\alpha_{nD} + \alpha_{nC})/2]}$ where ρ is the density of the gear material (for steel $\rho = 7,86 \cdot 10^{-6} \text{ kg/mm}^3$)	0,010 kg/mm	E: 14		
Relative wheel mass per unit facewidth reduced to the line of action	$m_2^* = \frac{1}{8} \cdot \rho \cdot \pi \cdot \frac{d_{m2}^2}{\cos^2[(\alpha_{nD} + \alpha_{nC})/2]}$ where ρ is the density of the gear material (for steel $\rho = 7,86 \cdot 10^{-6} \text{ kg/mm}^3$)	0,078 kg/mm	E: 14		
Mass reduced to the line of action of the dynamically equivalent cylindrical gear pair	$m_{red} = \frac{m_1^* \cdot m_2^*}{m_1^* + m_2^*}$	0,009 kg/mm	E: 10		
Resonance speed of pinion	$n_{E1} = \frac{30 \cdot 10^3}{\pi \cdot z_1} \cdot \sqrt{\frac{c_{\gamma}}{m_{red}}}$	32 314 min^{-1}	E: 9		
Dimensionless reference speed	$N = \frac{n_1}{n_{E1}}$	0,037	E: 8		

ISO/TR 10300-30:2024(E)

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
For virtual contact ratio, $\varepsilon_{v\gamma} = 2,392 > 2$	$c_{v1,2} = c_{v1} + c_{v2}$	0,593	T: 3		
	c_{v3}	0,115	T: 3		
	c_{v4}	0,473	T: 3		
	$c_{v5,6}$	0,654	T: 3		
	c_{v7}	0,993	T: 3		
Constant for the dynamic factor with $K_A = 1,1$ as given in Table A.4.	$K = \frac{b_v \cdot f_{p,eff} \cdot c'}{F_{vmt} \cdot K_A} \cdot c_{v1,2} + c_{v3}$	0,537	E: 16		
Dynamic factor	$K_{v-B} = N \cdot K + 1$	1,020	E: 15		
Determination of the length of contact lines					
For virtual contact ratio, $\varepsilon_{v\beta} = 1,227, \varepsilon_{v\beta} \geq 1$	$f_t = +p_{vet} \cos \beta_{vb}$	9,486 mm	T: A.2		
	f_m	0,000 mm	T: A.2		
	$f_r = -p_{vet} \cos \beta_{vb}$	-9,486 mm	T: A.2		
Maximum distances from middle contact line	$f_{\max B} = \frac{1}{2} \cdot [g_{v\alpha} + b_{v,eff} \cdot (\tan \gamma + \tan \beta_{vb})] \cdot \cos \beta_{vb}$	11,344 mm	E: A.33		
	$f_{\max 0} = \frac{1}{2} \cdot [g_{v\alpha} - b_{v,eff} \cdot (\tan \gamma + \tan \beta_{vb})] \cdot \cos \beta_{vb}$	-0,293 mm	E: A.34		
	$f_{\max} = f_{\max B}$ since $f_{\max B} > f_{\max 0}$	11,344 mm			

ISO/TR 10300-30:2024(E)

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Theoretical length of contact line	$l_{b0} = \sqrt{(x_1 - x_2)^2 + (y_1 - y_2)^2}$	24,345 mm	E: A.32		
Theoretical length of middle contact line calculated with $f=f_m$ for contact stress as specified in ISO 10300-2:2023, 6.1	$x_1 = \frac{f \cdot \cos \beta_{vb} + \tan \beta_{vb} \cdot \left(f \cdot \sin \beta_{vb} + \frac{b_{v,eff}}{2} \right) + \frac{1}{2} \cdot (g_{v\alpha} + b_{v,eff} \cdot \tan \gamma)}{\tan \gamma + \tan \beta_{vb}}$	21,047 mm	E: A.29		
	$x_2 = \frac{f \cdot \cos \beta_{vb} + \tan \beta_{vb} \cdot \left(f \cdot \sin \beta_{vb} + \frac{b_{v,eff}}{2} \right) - \frac{1}{2} \cdot (g_{v\alpha} - b_{v,eff} \cdot \tan \gamma)}{\tan \gamma + \tan \beta_{vb}}$	0,543 mm	E: A.30		
	$y_1 = -x_1 \cdot \tan \beta_{vb} + f \cdot \cos \beta_{vb} + \tan \beta_{vb} \cdot \left(f \cdot \sin \beta_{vb} + \frac{b_{v,eff}}{2} \right)$	-6,560 mm	E: A.31		
	$y_2 = -x_2 \cdot \tan \beta_{vb} + f \cdot \cos \beta_{vb} + \tan \beta_{vb} \cdot \left(f \cdot \sin \beta_{vb} + \frac{b_{v,eff}}{2} \right)$	6,560 mm	E: A.31		
Correction factor	$C_{lb} = \sqrt{\left(1 - \left(\frac{f}{f_{max}} \right)^2 \right) \cdot \left(1 - \sqrt{\frac{b_{v,eff}}{b_v}} \right)^2}$	0,078	E: A.36		
Length of contact line	$l_b = l_{b0} (1 - C_{lb})$	22,442 mm	E: A.26		
Length of middle contact line	$l_{bm} = l_b$	22,442 mm			

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Load sharing factor (macropitting)					
Exponent for calculation of parabolic distribution of peak loads	e_{LS}	3		T: 5	
Related peak load	$p^* = 1 - \left(\frac{ f }{ f_{max} } \right)^{e_{LS}}$	—		E: 7 F: 2	
Related peak load at tip contact line	$p_t^* = 1 - \left(\frac{ f_t }{ f_{max} } \right)^{e_{LS}}$	0,415			
Related peak load at middle contact line	$p_m^* = 1 - \left(\frac{ f_m }{ f_{max} } \right)^{e_{LS}}$	1,000			
Related peak load at root contact line	$p_r^* = 1 - \left(\frac{ f_r }{ f_{max} } \right)^{e_{LS}}$	0,415			
Related area	$A^* = \frac{1}{4} p^* \cdot l_b \cdot \pi$	—		E: 8 F: 2	
Related area at tip contact line	$A_t^* = \frac{1}{4} p_t^* \cdot l_{bt} \cdot \pi$	1,277 mm			

ISO/TR 10300-30:2024(E)

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Related area at middle contact line	$A_m^* = \frac{1}{4} p_m^* \cdot l_{bm} \cdot \pi$	17,626 mm			
Related area at root contact line	$A_r^* = \frac{1}{4} p_r^* \cdot l_{br} \cdot \pi$	1,277 mm			
Load sharing factor	$Z_{LS} = \sqrt{\frac{A_m^*}{A_t^* + A_m^* + A_r^*}}$	0,935		E: 10	
Face load factors (Calculation according to Method C)					
Load distribution factor	$K_{H\beta-C} = 1,5 \cdot K_{H\beta-be}$	1,650	E: 28		
with	$K_{H\beta-be}$	1,100	T:4		
Load distribution factor	$K_{F\beta-C} = K_{H\beta-C} / K_{F0}$	1,650	E: 32		
with	K_{F0}	1,000	E: 33		
Transverse load factors (Calculation according to Method B)					
Determinant tangential force at mid facewidth on the pitch cone	$F_{mtH} = F_{vmt} \cdot K_A \cdot K_v \cdot K_{H\beta}$	20 225,0 N	E: 47		
Transverse load factors for bevel gear with virtual contact ratio $\varepsilon_{v\gamma} \geq 2,3$ and $\varepsilon_{v\gamma} > 2$	$K_{H\alpha-B}^* = K_{F\alpha-B}^* = 0,9 + 0,4 \cdot \sqrt{\frac{2 \cdot (\varepsilon_{v\gamma} - 1)}{\varepsilon_{v\gamma}}} \cdot \frac{c_\gamma \cdot (f_{pt} - y_\alpha)}{F_{mtH} / b_v}$	1,161	E: 48		
Relative hypoid offset	$a_{rel} = \frac{2 \cdot a }{d_{m2}}$	0,000	E: 41		

ISO/TR 10300-30:2024(E)

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Transverse load factors	$K_{H\alpha} = K_{F\alpha} = K_{H\alpha}^* - \frac{K_{H\alpha}^* - 1}{0,1} \cdot a_{rel}$	1,161	E: 40		

Table A.7 — Calculation of surface durability (macropitting)

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Z-factors					
Factors for calculation of mid-zone factor for $\varepsilon_{v\beta} = 1,227 \geq 1$	$F_1 = \varepsilon_{v\alpha}$	1,165		T: 4	
	$F_2 = \varepsilon_{v\alpha}$	1,165		T: 4	
Mid-zone factor	$Z_{M-B} = \frac{\tan \alpha_{vet}}{\sqrt{\left[\sqrt{\left(\frac{d_{va1}}{d_{vb1}} \right)^2 - 1} - F_1 \cdot \frac{\pi}{z_{v1}} \right] \cdot \left[\sqrt{\left(\frac{d_{va2}}{d_{vb2}} \right)^2 - 1} - F_2 \cdot \frac{\pi}{z_{v2}} \right]}}$	0,916		E: 6	
Elasticity factor	$Z_E = \sqrt{\frac{1}{\pi \cdot \left[\frac{(1-\nu_1^2)}{E_1} + \frac{(1-\nu_2^2)}{E_2} \right]}}$	191,646 (N/mm ²) ^{1/2}		E: 51	
Bevel gear factor	Z_{KP}	1,2		E: 11	
Lubricant factor	$C_{ZL} = 0,08 \cdot \frac{\sigma_{H,lim} - 850}{350} + 0,83$ using $\sigma_{Hlim} = 1\ 200\ \text{N/mm}^2$, which is the upper allowed limit in the above formula.	0,910		E: 55	
	$Z_L = C_{ZL} + \frac{4 \cdot (1,0 - C_{ZL})}{\left(1,2 + \frac{134}{\nu_{40}} \right)^2}$	0,992		E: 54	

ISO/TR 10300-30:2024(E)

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Speed factor	$C_{ZV} = 0,08 \cdot \frac{\sigma_{H,lim} - 850}{350} + 0,85$ using $\sigma_{Hlim} = 1\,200\text{ N/mm}^2$, which is the upper allowed limit in the above formula.	0,930		E: 57	
	$Z_v = C_{ZV} + \frac{2 \cdot (1,0 - C_{ZV})}{\sqrt{0,8 + \frac{32}{v_{mt2}}}}$	0,974		E: 56	
Roughness factor with the radius of relative curvature $\rho = \rho_{rel} = 12,459\text{ mm}$ for Method B1 (see ISO 10300-1:2023, Annex A)	$Rz_{10} = \frac{Rz_1 + Rz_2}{2} \cdot \sqrt[3]{\frac{10}{\rho}}$	7,435 μm		E: 58	
	$C_{ZR} = 0,12 + \frac{1\,000 - \sigma_{H,lim}}{5\,000}$ using $\sigma_{Hlim} = 1\,200\text{ N/mm}^2$, which is the upper allowed limit in the above formula.	0,080		E: 60	
	$Z_R = \left(\frac{3}{Rz_{10}} \right)^{C_{ZR}}$	0,930		E: 59	
Product of the lubricant influence factors	$Z_L Z_v Z_R$	0,899			
Size factor	Z_x for Method B1 (see ISO 10300-2:2023, 6.5.2)	1,000			
Hypoid factor	Z_{Hyp} <i>Set $Z_{Hyp} = 1,0$ for non-offset.</i>	1,000		E: 12	
Life factor for pinion	$Z_{NT,1}$	1,000		T: 6	
Life factor for wheel	$Z_{NT,2}$	1,000		T: 6	

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Work hardening factor	Set $Z_{W,1} = Z_{W,2} = 1,0$ because pinion and wheel with equal hardness.	1,000		E: 70	
Contact stress formula					
Nominal normal force of the virtual cylindrical gear at mean point P	$F_n = \frac{F_{mt1}}{\cos \alpha_{nD} \cdot \cos \beta_{m1}}$ NOTE ISO 10300-2:2023, Formula (3) contains a misprint. The correct formula has been applied to this sample calculation.	14 193,3 N		E: 3	
Nominal value of the contact stress	$\sigma_{H0-B1} = \sqrt{\frac{F_n}{l_{bm} \cdot \rho_{rel}}} Z_{M-B} \cdot Z_{LS} \cdot Z_E$	1 168,5 N/mm ²		E: 2	
Contact stress	$\sigma_{H-B1} = \sigma_{H0-B1} \cdot \sqrt{K_A \cdot K_v \cdot K_{H\beta} \cdot K_{H\alpha}}$	1 712,8 N/mm ²		E: 1	
Permissible contact stress	$\sigma_{HP-B1} = \sigma_{H,lim} \cdot Z_{NT} \cdot Z_X \cdot Z_L \cdot Z_v \cdot Z_R \cdot Z_W \cdot Z_{KP} \cdot Z_{Hyp}$	1 617,8 N/mm ²		E: 4	
Calculated safety factor for contact stress (macropitting) on pinion and wheel	$S_{H1,2-B1} = \frac{\sigma_{HP1,2-B1}}{\sigma_{H1,2-B1}}$	0,944		E: 5	

Table A.8 — Calculation of tooth root strength for pinion

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Y-factors					
Load sharing factor	$Y_{LS} = Z_{LS}^2$	0,873			E: 36
Geometry values for pinion according to Tables A.2 and A.3.	$h_{a0} = h_{fp} = k_{hfp} m_{mn}$	4,017 mm			F: 2
Parameters for pinion	$E_{1,D} = \left(\frac{\pi}{4} - x_{sm1} \right) \cdot m_{mn} - h_{a0} \cdot \tan \alpha_{eD} - \frac{\rho_{a01,D} \cdot (1 - \sin \alpha_{eD}) - s_{pr1,D}}{\cos \alpha_{eD}}$	0,385 mm			E: 7
	$E_{1,C} = \left(\frac{\pi}{4} - x_{sm1} \right) \cdot m_{mn} - h_{a0} \cdot \tan \alpha_{eC} - \frac{\rho_{a01,C} \cdot (1 - \sin \alpha_{eC}) - s_{pr1,C}}{\cos \alpha_{eC}}$	0,385 mm			E: 7
	$G_{1,D} = \frac{\rho_{a01,D}}{m_{mn}} - \frac{h_{a0}}{m_{mn}} + x_{hm1}$	-0,496			E: 8
	$G_{1,C} = \frac{\rho_{a01,C}}{m_{mn}} - \frac{h_{a0}}{m_{mn}} + x_{hm1}$	-0,496			E: 8
	$H_{1,D} = \frac{2}{z_{vn1,D}} \cdot \left(\frac{\pi}{2} - \frac{E_{1,D}}{m_{mn}} \right) - \frac{\pi}{3}$	-0,934			E: 9
	$H_{1,C} = \frac{2}{z_{vn1,C}} \cdot \left(\frac{\pi}{2} - \frac{E_{1,C}}{m_{mn}} \right) - \frac{\pi}{3}$	-0,934			E: 9

ISO/TR 10300-30:2024(E)

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Iteration starting with $\vartheta = \pi/6$ until $(\vartheta_{\text{new}} - \vartheta) < 0,000\,001$	$\vartheta_{1,D,C} = \frac{2 \cdot G_{1,D,C}}{z_{vn1,D,C}} \cdot \tan \vartheta_{1,D,C} - H_{1,D,C}$	Initial: 52,221°			E: 10
		Final: 50,783°			E: 10
Tooth root chordal thickness on the drive side	$s_{Fn1,D} = m_{mn} \cdot z_{vn1,D} \cdot \sin\left(\frac{\pi}{3} - \vartheta_{1,D}\right) + m_{mn} \cdot \sqrt{3} \cdot \left(\frac{G_{1,D}}{\cos \vartheta_{1,D}} - \frac{\rho_{a01,D}}{m_{mn}}\right)$	7,423 mm			E: 11
Tooth root chordal thickness on coast side	$s_{Fn1,C} = m_{mn} \cdot z_{vn1,C} \cdot \sin\left(\frac{\pi}{3} - \vartheta_{1,C}\right) + m_{mn} \cdot \sqrt{3} \cdot \left(\frac{G_{1,C}}{\cos \vartheta_{1,C}} - \frac{\rho_{a01,C}}{m_{mn}}\right)$	7,423 mm			E: 11
Tooth root chord	$s_{Fn1} = 0,5 \cdot s_{Fn1,D} + 0,5 \cdot s_{Fn1,C}$	7,423 mm			E: 12
Fillet radius at contact point of 30° tangent on the drive side	$\rho_{F1,D} = \rho_{a01,D} + \frac{2 \cdot G_{1,D}^2 \cdot m_{mn}}{\cos \vartheta_{1,D} \cdot \left(z_{vn1,D} \cdot (\cos \vartheta_{1,D})^2 - 2 \cdot G_{1,D}\right)}$	1,023 mm			E: 13
Normal pressure angle at tooth tip on the drive side	$\alpha_{an1,D} = \arccos\left(\frac{d_{vbn1,D}}{d_{van1,D}}\right)$	32,776°			E: 16
Auxiliary angle for tooth form and tooth correction factor on the drive side	$\gamma_{a1,D} = \frac{1}{z_{vn1,D}} \cdot \left[\frac{\pi}{2} + 2(x_{hm1} \cdot \tan \alpha_{eD} + x_{sm1}) \right] + \text{inv} \alpha_{eD} - \text{inv} \alpha_{an1,D}$	1,242°			E: 17

ISO/TR 10300-30:2024(E)

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Load application angle at tooth tip of virtual cylindrical gear on drive side	$\alpha_{Fan1,D} = \alpha_{an1,D} - \gamma_{a1,D}$	31,534°			E: 15
Bending moment arm on the drive side	$h_{Fa1,D} = \frac{m_{mn}}{2} \cdot \left[\left(\cos \gamma_{a1,D} - \sin \gamma_{a1,D} \cdot \tan \alpha_{Fan1,D} \right) \cdot \frac{d_{van1,D}}{m_{mn}} - z_{vn1,D} \cdot \cos \left(\frac{\pi}{3} - \vartheta_{1,D} \right) - \frac{G_{1,D}}{\cos \vartheta_{1,D}} + \frac{\rho_{a01,D}}{m_{mn}} \right]$	6,405 mm			E: 14
Tooth form factor on the drive side	$Y_{Fa1,D} = \frac{6 \cdot \frac{h_{Fa1,D}}{m_{mn}} \cdot \cos \alpha_{Fan1,D}}{\left(\frac{s_{Fn1}}{m_{mn}} \right)^2 \cdot \cos \alpha_{n1,D}}$	2,033			E: 6
Stress correction factor on the drive side	$L_{a1,D} = \frac{s_{Fn1}}{h_{Fa1,D}}$	1,159			E: 25
	$q_{s1,D} = \frac{s_{Fn1}}{2 \rho_{F1,D}}$	3,630			E: 26
	$Y_{Sa1,D} = \left(1,2 + 0,13 \cdot L_{a1,D} \right) \cdot q_{s1,D}^{\left(\frac{1}{1,21+2,3/L_{a1,D}} \right)}$	2,022			E: 24
Relative surface condition factor	$Y_{R,relT} = \frac{Y_R}{Y_{RT}} = 1,674 - 0,529 \cdot (Rz + 1)^{1/10}$	0,972			E: 40

ISO/TR 10300-30:2024(E)

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
for $R_z = 16 \mu\text{m}$ and through hardened and case hardened steels					
Life factor for pinion	$Y_{NT,1}$	1,000			T: 4
Stress correction factor	Y_{ST}	2,000			E: 4
Size factor	$Y_X = 1,05 - 0,01m_{mn}$ Set $Y_X = 1,0$ since range is $0,8 \leq Y_X \leq 1,0$.	1,000			E: 209
Bevel spiral angle factor using l_{bm} according to Table A.6 for l_{bb}	$Y_{BS} = \frac{a_{BS}}{c_{BS}} \left(\frac{l_{bb}}{b_a} - 1,05 \cdot b_{BS} \right)^2 + 1$	1,054			E: 30
with	$b_a = b_v / \cos \beta_v$	31,008 mm			E: 34
	$l_{bb} = l_{bm} \frac{\cos \beta_{vb}}{\cos \beta_v}$	23,077 mm			E: 35
Relative notch sensitivity factor	$Y_{\delta,relT1} = \frac{1 + \sqrt{\rho' \cdot \chi_1^X}}{1 + \sqrt{\rho' \cdot \chi_T^X}}$	1,010			E: 43
Contact ratio factor, Y_ϵ for $\epsilon_{vs} = 1,227 > 1$	Y_ϵ	0,625			E: 29
Tooth root stress formula for pinion					

ISO/TR 10300-30:2024(E)

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Nominal tooth root stress	$\sigma_{F01-B1} = \frac{F_{vmt}}{b_v \cdot m_{mn}} \cdot Y_{Fa} \cdot Y_{Sa} \cdot Y_{\varepsilon} \cdot Y_{BS} \cdot Y_{LS}$	316,5 N/mm ²			E: 2
Local tooth root stress	$\sigma_{F1-B1} = \sigma_{F01-B1} \cdot K_A \cdot K_v \cdot K_{F\beta} \cdot K_{F\alpha}$	680,1 N/mm ²			E: 1
Permissible tooth root stress	$\sigma_{FP1-B1} = \sigma_{F,lim} \cdot Y_{ST} \cdot Y_{NT} \cdot Y_{\delta,relT-B1} \cdot Y_{R,relT-B1} \cdot Y_X$	942,0 N/mm ²			E: 4
Calculated safety for tooth root strength on pinion	$S_{F1-B1} = \frac{\sigma_{FP1-B1}}{\sigma_{F1-B1}}$	1,385			E: 5

Table A.9 — Calculation of tooth root strength for wheel

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Y-factors different from pinion					
Geometry values for wheel according to Tables A.2 and A.3.	$h_{a0} = h_{fp} = k_{hfp} m_{mn}$	4,017 mm			F: 2
Parameters for wheel	$E_{2,D} = \left(\frac{\pi}{4} - x_{sm2} \right) \cdot m_{mn} - h_{a0} \cdot \tan \alpha_{eD} - \frac{\rho_{a02,D} \cdot (1 - \sin \alpha_{eD}) - s_{pr2,D}}{\cos \alpha_{eD}}$	0,398 mm			E: 7
	$E_{2,C} = \left(\frac{\pi}{4} - x_{sm2} \right) \cdot m_{mn} - h_{a0} \cdot \tan \alpha_{eC} - \frac{\rho_{a02,C} \cdot (1 - \sin \alpha_{eC}) - s_{pr2,C}}{\cos \alpha_{eC}}$	0,398 mm			E: 7
	$G_{2,D} = \frac{\rho_{a02,D}}{m_{mn}} - \frac{h_{a0}}{m_{mn}} + x_{hm2}$	-1,382			E: 8
	$G_{2,C} = \frac{\rho_{a02,C}}{m_{mn}} - \frac{h_{a0}}{m_{mn}} + x_{hm2}$	-1,382			E: 8
	$H_{2,D} = \frac{2}{z_{vn2,D}} \cdot \left(\frac{\pi}{2} - \frac{E_{2,D}}{m_{mn}} \right) - \frac{\pi}{3}$	-1,033			E: 9
	$H_{2,C} = \frac{2}{z_{vn2,C}} \cdot \left(\frac{\pi}{2} - \frac{E_{2,C}}{m_{mn}} \right) - \frac{\pi}{3}$	-1,033			E: 9

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Iteration starting with $\vartheta = \pi / 6$ until $(\vartheta_{\text{new}} - \vartheta) < 0,000\,001$	$\vartheta_{2,D,C} = \frac{2 \cdot G_{2,D,C}}{z_{vn2,D,C}} \cdot \tan \vartheta_{2,D,C} - H_{2,D,C}$	Initial: 58,705°			E: 10
		Final: 57,895°			E: 10
Tooth root chordal thickness on the drive side	$s_{Fn2,D} = m_{mn} \cdot z_{vn2,D} \cdot \sin\left(\frac{\pi}{3} - \vartheta_{2,D}\right) + m_{mn} \cdot \sqrt{3} \cdot \left(\frac{G_{2,D}}{\cos \vartheta_{2,D}} - \frac{\rho_{a02,D}}{m_{mn}}\right)$	6,898 mm			E: 11
Tooth root chordal thickness on coast side	$s_{Fn2,C} = m_{mn} \cdot z_{vn2,C} \cdot \sin\left(\frac{\pi}{3} - \vartheta_{2,C}\right) + m_{mn} \cdot \sqrt{3} \cdot \left(\frac{G_{2,C}}{\cos \vartheta_{2,C}} - \frac{\rho_{a02,C}}{m_{mn}}\right)$	6,898 mm			E: 11
Tooth root chord	$s_{Fn2} = 0,5 \cdot s_{Fn2,D} + 0,5 \cdot s_{Fn2,C}$	6,898 mm			E: 12
Fillet radius at contact point of 30° tangent	$\rho_{F2,D} = \rho_{a02,D} + \frac{2 \cdot G_{2,D}^2 \cdot m_{mn}}{\cos \vartheta_{2,D} \cdot \left(z_{vn2,D} \cdot (\cos \vartheta_{2,D})^2 - 2 \cdot G_{2,D}\right)}$	1,592 mm			E: 13
Normal pressure angle at tooth tip on the drive side	$\alpha_{an2,D} = \arccos\left(\frac{d_{vbn2,D}}{d_{van2,D}}\right)$	20,766°			E: 16
Auxiliary angle for tooth form and tooth correction factor on the drive side	$\gamma_{a2,D} = \frac{1}{z_{vn2,D}} \cdot \left[\frac{\pi}{2} + 2(x_{hm2} \cdot \tan \alpha_{eD} + x_{sm2}) \right] + \text{inv} \alpha_{eD} - \text{inv} \alpha_{an2,D}$	0,209°			E: 17

ISO/TR 10300-30:2024(E)

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Load application angle at tooth tip of virtual cylindrical gear on the drive side	$\alpha_{\text{Fan2,D}} = \alpha_{\text{an2,D}} - \gamma_{\text{a2,D}}$	20,557°			E: 15
Bending moment arm on the drive side	$h_{\text{Fa2,D}} = \frac{m_{\text{mn}}}{2} \cdot \left[(\cos \gamma_{\text{a2,D}} - \sin \gamma_{\text{a2,D}} \cdot \tan \alpha_{\text{Fan2,D}}) \cdot \frac{d_{\text{van2,D}}}{m_{\text{mn}}} - z_{\text{vn2,D}} \cdot \cos \left(\frac{\pi}{3} - \vartheta_{2,D} \right) - \frac{G_{2,D}}{\cos \vartheta_{2,D}} + \frac{\rho_{\text{a02,D}}}{m_{\text{mn}}} \right]$	6,141 mm			E: 14
Tooth form factor on the drive side	$Y_{\text{Fa2,D}} = \frac{6 \cdot \frac{h_{\text{Fa2,D}}}{m_{\text{mn}}} \cdot \cos \alpha_{\text{Fan2,D}}}{\left(\frac{s_{\text{Fn2}}}{m_{\text{mn}}} \right)^2 \cdot \cos \alpha_{\text{n2,D}}}$	2,480			E: 6
Stress correction factor on the drive side	$L_{\text{a2,D}} = \frac{s_{\text{Fn2}}}{h_{\text{Fa2,D}}}$	1,123			E: 25
	$q_{\text{s2,D}} = \frac{s_{\text{Fn2}}}{2\rho_{\text{F2,D}}}$	2,166			E: 26
	$Y_{\text{Sa2,D}} = (1,2 + 0,13 \cdot L_{\text{a2,D}}) \cdot q_{\text{s2,D}}^{\left(\frac{1}{1,21+2,3/L_{\text{a2,D}}} \right)}$	1,706			E: 24
Relative surface condition factor for $Rz \leq 10 \mu\text{m}$ for through	$Y_{\text{R,relT}} = \frac{Y_{\text{R}}}{Y_{\text{RT}}} = 1,674 - 0,529 \cdot (Rz + 1)^{1/10}$	0,972			E: 40

ISO/TR 10300-30:2024(E)

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
hardened and case hardened steels					
Bevel spiral angle factor using l_{bm} according to Table A.6 for l_{bb}	$Y_{BS} = \frac{a_{BS}}{c_{BS}} \cdot \left(\frac{l_{bb}}{b_a} - 1,05 \cdot b_{BS} \right)^2 + 1$	1,054			E: 30
where	$b_a = b_v / \cos \beta_v$	31,008 mm			E: 34
	$l_{bb} = l_{bm} \cdot \frac{\cos \beta_{vb}}{\cos \beta_v}$	23,077 mm			E: 35
Relative notch sensitivity factor	$Y_{\delta,relT2} = \frac{1 + \sqrt{\rho' \cdot \chi_2^X}}{1 + \sqrt{\rho' \cdot \chi_T^X}}$	0,997			E: 43
Tooth root stress formula for wheel					
Nominal tooth root stress	$\sigma_{F02-B1} = \frac{F_{vmt}}{b_v \cdot m_{mn}} \cdot Y_{Fa} \cdot Y_{Sa} \cdot Y_{\varepsilon} \cdot Y_{BS} \cdot Y_{LS}$	325,9 N/mm ²			E: 2
Local tooth root stress	$\sigma_{F2-B1} = \sigma_{F02-B1} \cdot K_A \cdot K_v \cdot K_{F\beta} \cdot K_{F\alpha}$	700,1 N/mm ²			E: 1
Permissible tooth root stress	$\sigma_{FP2-B1} = \sigma_{F,lim} \cdot Y_{ST} \cdot Y_{NT} \cdot Y_{\delta,relT2-B1} \cdot Y_{R,relT-B1} \cdot Y_X$	929,8 N/mm ²			E: 4
Calculated safety for tooth root strength on wheel	$S_{F2-B1} = \frac{\sigma_{FP2-B1}}{\sigma_{F2-B1}}$	1,328			E: 5

A.3 Calculation of Sample 1 according to Method B2

Table A.10 — Calculation of surface durability (macropitting)

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Approximate values for application factors					
Relative facewidth	$b_{v,rel} = \frac{b_2}{m_{et2}}$	5,600	E: B.1		
Relative mean back cone distance on pinion	$R_{mpt1} = \frac{R_{m1} \cdot \tan \delta_1}{m_{et2}}$	6,432	E: B.2		
Relative mean back cone distance on wheel	$R_{mpt2} = \frac{R_{m2} \cdot \tan \delta_2}{m_{et2}}$	49,915	E: B.2		
Face contact ratio for bevel gears	$\varepsilon_{v\beta} = b_2 \cdot \sin \beta_{m2} / (\pi \cdot m_{mn})$	1,443	E: B.4		
Relative mean virtual pitch radius on pinion	$r_{vn1} = \frac{R_{mpt1}}{\cos^2 \beta_{m1}}$	9,586	E: B.6		
Relative mean virtual pitch radius on wheel	$r_{vn2} = \frac{R_{mpt2}}{\cos^2 \beta_{m2}}$	74,388	E: B.6		
Relative centre distance	$a_{vn} = r_{vn1} + r_{vn2}$	83,974	E: B.7		
Relative mean virtual dedendum on pinion	$h_{vfm1} = h_{fm1} / m_{et2}$	0,528	E: B.8		

ISO/TR 10300-30:2024(E)

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Relative mean virtual dedendum on wheel	$h_{vfm2} = h_{fm2} / m_{et2}$	1,244	E: B.8		
Relative virtual tooth thickness on pinion	$s_{vmn1} = s_{mn1} / m_{et2}$	1,425	E: B.9		
Relative virtual tooth thickness on wheel	$s_{vmn2} = s_{mn2} / m_{et2}$	0,774	E: B.9		
Relative mean virtual tip radius on pinion	$r_{va1} = r_{vn1} + h_{am1} / m_{et2}$	10,652	E: B.10		
Relative mean virtual tip radius on wheel	$r_{va2} = r_{vn2} + h_{am2} / m_{et2}$	74,739	E: B.10		
Angular pitch of virtual cylindrical wheel (required in ISO 10300-3:2023, 7.4.5)	$\theta_{v2} = \frac{\pi \cdot m_{mn}}{m_{et2} \cdot r_{vn2}}$	1,714°	E: B.11		
Relative edge radius of tool on pinion	$\rho_{va01} = \rho_{a01} / m_{et2}$	0,176	E: B.12		
Relative edge radius of tool on wheel	$\rho_{va02} = \rho_{a02} / m_{et2}$	0,265	E: B.12		
Virtual spiral angle for bevel gears without hypoid offset	$\beta_v = \beta_{m2}$	35°	E: B.13		
Adjusted pressure angle	$\alpha_a = \alpha_{eD} - \frac{\pi}{2} \cdot \cos \delta_2 \cdot \cos \beta_{m2} / z_2$	19,361°	E: B.16		

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Base virtual helix angle	$\sin \beta_{vb} = \sin \beta_v \cdot \cos \alpha_a$	$\sin(32,761^\circ)$	E: B.17		
Relative mean virtual base radius on pinion	$r_{vbn1} = r_{vn1} \cdot \cos \alpha_a$	9,044	E: B.18		
Relative mean virtual base radius on wheel	$r_{vbn2} = r_{vn2} \cdot \cos \alpha_a$	70,182	E: B.18		
Relative length of action from pinion tip to pitch circle in the normal section	$g_{v\alpha na} = \sqrt{r_{va1}^2 - r_{vbn1}^2} - r_{vn1} \cdot \sin \alpha_a$	2,451	E: B.19		
Relative length of action from wheel tip to pitch circle in the normal section	$g_{v\alpha nr} = \sqrt{r_{va2}^2 - r_{vbn2}^2} - r_{vn2} \cdot \sin \alpha_a$	1,038	E: B.20		
Relative length of action in normal section	$g_{v\alpha n} = g_{v\alpha na} + g_{v\alpha nr}$	3,489	E: B.21		
Relative mean normal pitch of virtual cylindrical gear	$p_{mn} = \frac{2,0 \cdot \pi \cdot m_{mn}}{m_{et2} \cdot \cos \alpha_a \cdot (\cos^2 \beta_{m1} + \cos^2 \beta_{m2} + 2,0 \cdot \tan^2 \alpha_a)}$	2,969	E: B.22		
Profile contact ratio in mean normal section	$\varepsilon_{v\alpha n} = \frac{g_{v\alpha n}}{p_{mn}}$	1,175	E: B.23		
Profile contact ratio in mean transverse section	$\varepsilon_{v\alpha} = \varepsilon_{v\alpha n} \cdot (\cos \beta_{vb})^2$	0,831	E: B.24		

ISO/TR 10300-30:2024(E)

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Modified contact ratio for bevel gears without hypoid offset	$\varepsilon_{v\gamma} = \sqrt{\varepsilon_{v\alpha}^2 + \varepsilon_{v\beta}^2}$	1,665	E: B.25		
Contact shift factor; see also ISO 10300-1:2023, Figure B.7.	$k' = \frac{z_2 - z_1}{3,2 \cdot z_2 + 4,0 \cdot z_1}$	0,138	E: B.26		
Macropitting resistance geometry factor					
Angle between contact direction and tooth tangent in pitch plane	$\cot(\beta_{m1} - \lambda_1) = \frac{\cos \zeta_{mp}}{\cos \beta_{m1} \cdot \cos \beta_{m2} \cdot \tan(\beta_{m1} - \lambda_r)} - \tan \beta_{m2}$	$\cot(35^\circ)$		E: 24	
Angle between projection of pinion axis and direction of contact in pitch plane	$\lambda_1 = \beta_{m1} - (\beta_{m1} - \lambda_1)$	0°		E: 25	
Angle of contact line relative to the root cone	$\tan w = \frac{\sin \alpha_a \cdot \tan(\beta_{m1} - \lambda_r)}{\cos \alpha_{lim}}$	$\tan(13,069^\circ)$		E: 26	
Mean base spiral angle	$\cos \beta_{bm} = \frac{1,0}{\sqrt{\tan^2(\beta_{m1} - \lambda_r) \cdot \cos^2 \alpha_a + 1,0}}$	$\cos(33,449^\circ)$		E: 27	
Relative mean normal base pitch	$p_{nb} = \frac{\pi \cdot m_{mn} \cdot \cos \alpha_a \cdot \cos \beta_{bm}}{m_{et2} \cdot \cos(\beta_{m1} - \lambda_r)}$	2,139		E: 28	
Angle between projection of wheel	$\lambda_2 = (\beta_{m1} - \lambda_r) - \beta_{m2}$	0°		E: 29	

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
axis and direction of contact in pitch plane					
Relative base facewidth	$b_b = \frac{b_2}{m_{et2} \cdot \cos \lambda_2}$	5,600		E: 30	
Pressure angle at point of load application on pinion	$\cos \alpha_{L1} = \cos \alpha_a \cdot \left[1, 0 - \frac{(r_{va1} - r_{vn1}) \cdot \cos^2 \beta_{m1}}{r_{va1} - r_{vn1} + R_{mpt1}} \right]$	cos(31,415°)		E: 31	
Pressure angle at point of load application on wheel	$\cos \alpha_{L2} = \cos \alpha_a \cdot \left[1, 0 - \frac{(r_{va2} - r_{vn2}) \cdot \cos^2 \beta_{m2}}{r_{va2} - r_{vn2} + R_{mpt2}} \right]$	cos(20,110°)		E: 31	
Radius of curvature difference between point of load application and mean point on pinion	$\rho_{\Delta 1} = \frac{r_{va1} - r_{vn1} + R_{mpt1}}{(\cos \beta_{m1})^2} \cdot \cos \alpha_{L1} \cdot (\tan \alpha_{L1} - \tan \alpha_a)$	2,474		E: 32	
Radius of curvature difference between point of load application and mean point on wheel	$\rho_{\Delta 2} = \frac{r_{va2} - r_{vn2} + R_{mpt2}}{(\cos \beta_{m2})^2} \cdot \cos \alpha_{L2} \cdot (\tan \alpha_{L2} - \tan \alpha_a)$	1,038		E: 32	
Radius of curvature change	$\rho_{\Delta red} = \cos \beta_{bm} \cdot (\rho_{\Delta 1} + \rho_{\Delta 2})$	2,930		E: 33	

ISO/TR 10300-30:2024(E)

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula, T: Table, F: Figure		
Relative length of action within the contact ellipse	$g_{\eta} = \sqrt{\rho_{\Delta\text{red}}^2 \cdot (\cos \beta_{\text{bm}})^2 + b_{\text{b}}^2 \cdot (\sin \beta_{\text{bm}})^2}$	3,937		E: 34	
Radius of relative profile curvature and load sharing ratio at critical point					
Length of action at critical point in contact ellipse (iteration starting with $y_1 = 0$)	$g_{\eta\text{I}} = \sqrt{g_{\eta}^2 - 4,0 y_1^2}$	Initial: 3,937		E: 36	
		Final: 3,920		E: 36	
Length of action considering adjacent teeth	$g_{\eta\text{I}\Sigma}^3 = g_{\eta\text{I}}^3 + \sum_{k=1}^{k=x} \sqrt{[g_{\eta\text{I}}^2 - 4 \cdot k \cdot p_{\text{nb}} \cdot (k \cdot p_{\text{nb}} + 2 \cdot y_1)]^3} + \sum_{k=1}^{k=y} \sqrt{[g_{\eta\text{I}}^2 - 4 \cdot k \cdot p_{\text{nb}} \cdot (k \cdot p_{\text{nb}} - 2 \cdot y_1)]^3}$	Initial: 61,046		E: 37	
		Final: 60,352		E: 37	
Load sharing ratio	$\varepsilon_{\text{NI}} = g_{\eta\text{I}}^3 / g_{\eta\text{I}\Sigma}^3$	Initial: 1,000		E: 38	
		Final: 0,998		E: 38	
Length of contact line	$g_{\text{c}} = g_{\eta\text{I}} \cdot \rho_{\Delta\text{red}} \cdot b_{\text{b}} / g_{\eta}^2$	Initial: 4,167		E: 39	
		Final: 4,148		E: 39	
Position change along path of contact	$g_{\eta\Delta} = \frac{\rho_{\Delta\text{red}}^2 \cdot y_1}{g_{\eta\text{I}}^2} + k' \cdot g_{\text{c}} \cdot \tan \beta_{\text{bm}} + \frac{0,5 \cdot \rho_{\Delta\text{red}}}{\cos \beta_{\text{bm}}} - \rho_{\Delta 2}$	Initial: 1,098		E: 40	
		Final: 0,993		E: 40	